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Waste heat recovery from a landfill gas-fired power plant

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ABSTRACT

Waste treatment and management is a certain challenge especially in areas with high population density. One of the options for waste treatment is landfilling, where the amount of municipal waste also produces landfill gas through anaerobic digestion. The heating value of the landfill gas is high enough to use it as a fuel in combustion processes, e.g. in internal combustion engines (ICEs) to produce electric power.

In Ano Liosia, Athens (Greece) up to 6000 tons of waste are landfilled every day and the landfill gas is used in an ICE power station directly at the site of the landfill. The power station consists of 15 ICEs and has an installed capacity of 23.5 MW. The major advantages of using ICE for power generation are the high electrical efficiency of ICEs and their fast load response. However, more than 50% of the landfill gas energy content is still released to the atmosphere as engine waste heat (exhaust gas and engine cooling water).

The aim of this paper is to study the possibilities of using this large amount of heat in order to increase the electricity production and efficiency of the Ano Liosia power station. Therefore, a thermodynamic and economic analysis of two different waste heat recovery (WHR) systems is conducted. The water/steam cycle and the Organic Rankine Cycle (ORC) are examined and evaluated by means of thermodynamic cycle simulation and by calculating their specific costs of power generation. Their advantages and disadvantages considering their application in landfillgas-fired ICE power stations are discussed under the consideration of maximal thermodynamic efficiency and minimal costs of power generation.

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Nomenclature

a	annuity (€/a)
A	area (m^2); cost (€)
c_{EP}	specific costs of electricity production (€/MW h)
d	inner fin diameter (m)
D	outer fin diameter (m)
$\dot{E}$	exergy flow (kW)
f_m	maintenance factor
k	overall heat transfer coefficient (kW/m^{2}K)
l	pipe length (m)
p	pressure (bar)
P	power (kW)
pp	pinch point (K)
Pr	Prandtl number
q	heat flux (kW/m^2); interest factor
$\dot{Q}$	heat flow (kW)
Nu	Nusselt number
Re	Reynolds number
s	fin thickness (m)
S	salary (€/a)
T	temperature (°C); time (a)

Greek symbols

α	heat transfer coefficient (kW/m^{2}K)
Δp	pressure loss (mbar)
η	efficiency
λ	heat transfer capacity (W/mK)

Subscripts and superscripts

c	cold side; capital
el	electrical
ex	exergetic; exhaust
gen	generator
h	hot side; hydraulic
l	life time
$mech$	mechanical
op	operational
P	pump
R	fin
s	steam; isentropic
T	turbine
th	thermal
w	water

1. Introduction

1.1. Motivation and purpose of the study

Due to the increasing amount of municipal waste – generated because of the growing population worldwide and the increased needs of consumption – especially in the developed countries, one of the major issues for every big city is the treatment of municipal waste. Many different treatment methods for waste management exist today. According to Eurostat [1], the treatment methods in the countries of the European Union vary from landfill and recycling to incineration and composting, depending on the economic status, the know-how and the legislation of every country.

As far as Greece is concerned, nearly the 80% of municipal waste is landfilled while the rest is recycled [1]. Referring to the region of Athens with a population of more than 4 million people, the waste treatment is a timeless problem. Nowadays, the majority of waste is deposited in the sanitary landfill of Ano Liosia, which is the biggest landfill of the country with a total area of more than 370,000 m^2 and receiving around 6000 tons of waste per day. The large amount of daily incoming waste ensures a stable production of landfill gas through the anaerobic digestion of the buried waste. This gas consists mainly of methane, which is one of the main greenhouse gases, being 21 times more potent than carbon dioxide concerning the global warming effect [2].

Taking advantage of this, the Greek company 'Helector' [3], in association with the Australian 'Energy Developments Ltd.' [4], constructed a power plant based on internal combustion engines (ICEs) next to the landfill. The station has a twofold role as it uses the landfill gas in order to produce electricity while preventing it from being released to the atmosphere.

However, a significant problem of the ICE power station is the large amount of waste heat generated by the ICEs mainly because of the engine exhaust gas, which is not further used for the generation of heat or power. At the present configuration, only the cooling water heat of some of the engines is used in order to operate the evaporators in a biological treatment unit. Therefore, the purpose of this study is to find a viable solution to utilize this high amount of wasted energy in order to generate electric power.

1.2. Literature

In literature, both the water/steam and the Organic Rankine Cycle have been considered as favorable bottoming cycles for the waste heat recovery (WHR) of internal combustion engines. Authors have made different approaches in the research of engine waste heat recovery. In case of the water/steam cycle, the research is focused on the optimization of the waste heat recovery system by means of thermal and exergetic analyses or parametric studies. But since the waste heat temperature level of ICEs is quite low (350–500 °C) compared to e.g. gas turbines, just a few studies are dealing with the optimized waste heat utilization of ICEs with water/steam cycles [5–7]. Quite a lot of studies can be found on Organic Rankine Cycles as bottoming cycles for ICEs. Different approaches of cycle and efficiency optimization are made in the studies found in literature:

- defining the optimal working fluid for WHR between two defined temperature levels (high cycle temperature = evaporation temperature, low cycle temperature = condensation temperature) [8–14]

- optimization of cycle configuration (use of internal heat exchanger, introduction of superheating, reheat and regenerative cycles) [15–18]
- economical evaluation of ORC cycles by calculating the size of important cycle components (heat exchangers, turbine) under the variation of important influence parameters (evaporation pressure and temperature) [13,19,20].

Studies on the technological and economical comparison between water/steam and ORC cycle or alternative technologies for certain WHR applications to define the most favorable cycle are almost missing in literature. Hountalas [21] and Woodward [22] are examining mechanical and electrical turbocompounding and Rankine bottoming cycles solely under thermodynamic considerations while economic aspects are also taken into account in [23].

Since the exhaust gas temperature level (about 400–500 °C) of the engines at the landfill is in a range where both cycles may have specific advantages in terms of thermal efficiency, power generation and economic considerations, it is impossible to define the best cycle configuration just on the basis of state of the art technology and literature survey. Therefore, the aim of this paper is to search for the optimal waste heat recovery system under thermodynamic and economical aspects by evaluating both the water/steam and the ORC technology.

At first both cycles will be examined on a thermodynamic basis using the software IPSEpro [24] and Excel for process simulations in order to find the optimal cycle configuration and working parameters. Additionally, a complete economic investigation by calculating the electricity production costs (based on offers by component manufacturers) is conducted in order to conclude to the most efficient solution. The result of this paper is the proposition of a viable system that could work alongside with the existing plant and contribute to further electricity production.

1.3. Description of the plant layout and operation

1.3.1. Plant construction and layout

The power plant was constructed in two phases. The first part of the station started its commercial operation in 2001. This part consists of 11 Deutz TBG 620 V16 K [25] power generation gensets of 1.255 MW each. By the end of 2006 the plant was expanded with the installation of four additional gensets, raising the total installed capacity up to 23.5 MW. This second part of the station consists of four General Electric Jenbacher 620 GS [26] power generation gensets of 2.433 MW each. All engines are operated fully autonomous and work continuously throughout the year.

Some important engine parameters which are necessary for system efficiency calculations are given in Table 1.

1.3.2. Landfill gas

The engines are fired with landfill gas. The gas composition is given in Table 2. The heating value of the landfill gas can be calculated according to [27] to 24.43 MJ/Nm^3.

One of the most important characteristics of this landfill gas is that it contains no sulfur either as pure element or in other compounds. Concerning this, acid dew point will not set any limitations in the following heat recovery analysis.

Table 1
Engine parameters.

	Deutz engine	Jenbacher engine
Nominal power	1255 kW	2433 kW
Nominal el. efficiency	40.2%/38.5%	43.0%[a]/39.7%[b]
Average load in operation	1008 kW	1944 kW
Average el. efficiency in operation	32.3%	32.2%

[a] For operation with natural gas.
[b] For operation with biogas/landfill gas.

Table 2
Gas composition of landfill gas.

Compound		Fraction (vol.%)
Methane	CH_4	48.5%
Carbon dioxide	CO_2	31.5%
Nitrogen	N_2	18.0%
Oxygen	O_2	1.5%
Hydrogen	H_2	<0.1%
Carbon monoxide	CO	<0.1%
Higher hydrocarbons	C_2H_6, C_3H_8, etc.	<0.1%

1.3.3. Plant operation

Regarding the operation of the plant, an excessive gas extraction rate could result to the destruction of the bacteria, which are responsible for the methanogenesis, and could thus stop the gas production. For that reason the operating company keeps a balance between the gas extraction and the working load of the engines. On the other hand, the continuous operation of the plant causes many malfunctions and damages to the engines. Consequently, mainly due to the frequent maintenance of the engines and the limitations in the landfill gas extraction, the availability factor of the plant is approximately 80%, while the daily energy production is about 440,000 kW h el. The maintenance of the engines is so scheduled, that throughout the year, the daily energy production is constant.

Apart from the landfill gas, another major pollutant product of the landfill is leachate. The operating company has constructed a leachate treatment unit next to the power plant in order to further reduce the environmental impact of the landfill to the surrounding area. For the operation of the final stage of the leachate treatment process, the waste heat from the cooling water of the four GE Jenbacher engines is used.

The waste heat of the engines' exhaust gases and of the Deutz engines' cooling water is not further used. This can be seen in the Sankey energy flow chart (see Fig. 1), which is showing the current operational condition of the power plant.

2. Cycle modelling and thermodynamic calculation

The modelling and thermodynamic calculation of the different bottoming cycles will be presented in this section. Formulations to derive energetic and exergetic efficiencies of the ICE power plant with Steam-Rankine or Organic-Rankine Cycle are presented and the conditions and restrictions for the calculations will be defined. Additionally, the principles of heat exchanger area calculation are presented in this section in order to analyse the Steam-Rankine Cycle configurations in terms of heat exchanger area. From the calculation of the heat exchanger area the investment costs of these heat exchangers can be derived, which is important for the economic evaluation of different cycles.

2.1. Energetic and exergetic evaluation

The thermodynamic efficiency η_{th} is calculated by means of the first law of thermodynamics [27]:

$$\eta_{th} = \frac{P_{el,net}}{\dot{Q}_{input}} \tag{1}$$

with

$$P_{el,net} = (P_{th,turbine} - P_{own\,consumption}) \cdot \eta_{mech} \cdot \eta_{gen} \tag{2}$$

$$\dot{Q}_{input} = \dot{Q}_{exhaust} + \dot{Q}_{HTcooling} \tag{3}$$

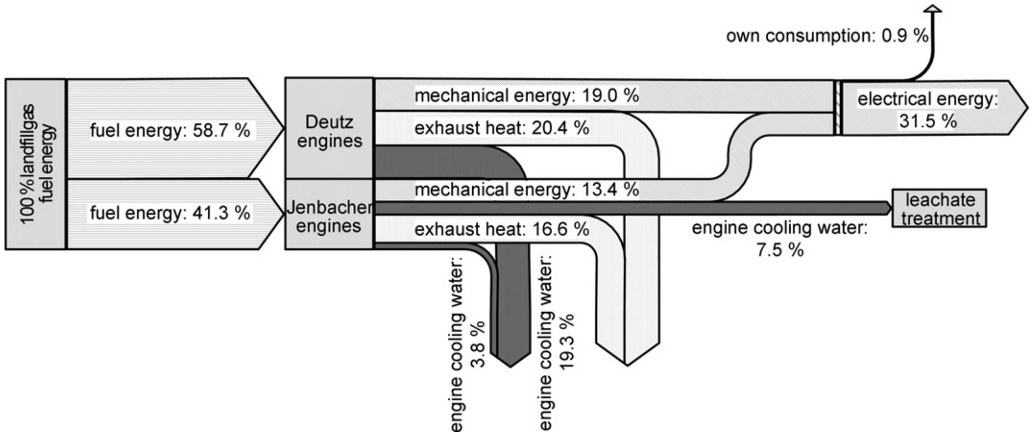

Fig. 1. Sankey energy flow chart of current plant operation.

In order to calculate the net electrical output $P_{el,net}$ and thermodynamic efficiency η_{th} of the WHR cycles, the mechanical and electrical efficiency of the various components are considered as in Table 3 [28].

The exergetic efficiency of the cycle and the components is evaluated according to the second law of thermodynamics. Formulas for the cycle exergetic efficiency:

$$\eta_{ex} = \frac{P_{el,net}}{\dot{E}_{input}} \tag{4}$$

with

$$\dot{E}_{input} = \dot{E}_{exhaust} + \dot{E}_{HTcooling} \tag{5}$$

$$\dot{E}_{ex/HT} = \dot{m}_{ex/HT} \cdot [h_{in} - h_0 - T_0 \cdot (s_{in} - s_0)] \tag{6}$$

Formulas for component efficiencies are given by Nikulshin et al. [29] and Mago and Charma [30]:

Heat exchanger: $\eta_{ex,HE} = \frac{\dot{E}_{c,out} - \dot{E}_{c,in}}{\dot{E}_{h,in} - \dot{E}_{h,out}}$

Turbine: $\eta_{ex,T} = \frac{P_{el,T}}{\dot{E}_{s,in} - \dot{E}_{s,out}}$

Pump: $\eta_{ex,P} = \frac{\dot{E}_{w,out} - \dot{E}_{w,in}}{P_{el,P}}$

2.2. Heat exchanger area calculation

In this section the principles of calculating the area of different types of heat exchangers within a Steam-Rankine Cycle are explained.

According to the principle of heat transfer, the area of a heat exchanger A can be determined from the amount of transferred heat $\dot{Q}$, the overall heat transfer coefficient k and the mean logarithmic temperature difference ΔT_{log} [31]:

$$\dot{Q} = k \cdot A \cdot \Delta T_{log} \tag{7}$$

In order to calculate the overall heat transfer coefficient k for pipes, the convection heat transfer coefficients for the inner and outer pipe surface α_i and α_o and the thermal conductivity of the pipe material λ are needed. The calculation of the convection heat transfer coefficients is based on Nusselt-correlations given by the VDI heat atlas [32]. The most important formulas to calculate the heat exchanger area within the HRSG are described here. The HRSG has to be divided into different sections (according to Table 4) with different flow regimes and therefore different convective heat transfer coefficients.

The HRSG consists of bundles of staggered, finned pipes. For staggered finned pipes the exhaust side heat transfer coefficient α_o can be determined by calculating the outer heat transfer coefficient for plain pipes α_R, using a Nusselt-correlation, and the fin efficiency η_R [32]:

$$\alpha_o = \alpha_R \left[1 - (1 - \eta_R)\frac{A_R}{A}\right] \tag{8}$$

$$\alpha_R = \frac{Nu \cdot \lambda_{ex}}{d} \tag{9}$$

$$Nu = 0.38 \cdot Re^{0.6} \cdot \left(\frac{A}{A_{G_0}}\right)^{-0.15} \cdot Pr^{1/3} \tag{10}$$

$$\eta_R = \frac{\tanh X}{X} \tag{11}$$

$$X = \phi\frac{d}{2}\sqrt{\frac{2\alpha_R}{\alpha_R \cdot s}} \tag{12}$$

$$\phi = \frac{D}{d-1}\left[1 + 0.35 \ln \frac{D}{d}\right] \tag{13}$$

The inner heat transfer coefficient α_i can be determined by using the equation of Dittus–Bölter for single phase water or steam flow (economizer and superheater) [31]:

$$Nu = 0.024 \cdot Re^{0.8} \cdot Pr^{1/3} \cdot \left(1 + \frac{d_h}{l}\right)^{2/3} \tag{14}$$

In the case of the evaporator the inner heat transfer coefficient α_i for saturated water steam flow is not directly calculated. The area calculation of the evaporator is based on a pipe wise iteration over the steam content x, the inner pipe wall temperature T_{wall} and the heat flux to the inner pipe wall q_{wall}. According to Thom et al. [33]

Table 3
Efficiencies for WHR calculations.

		CRC	ORC
Isentropic turbine efficiency	$\eta_{is,T}$	75%	85%
Isentropic pump efficiency	$\eta_{is,P}$	80%	80%
Mechanical efficiency	η_{mech}	98%	98%
Generator efficiency	η_{gen}	98%	98%

Table 4
Types of heat exchangers within a Steam-Rankine Cycle.

HRSG section	Inner fluid flow	Outer fluid flow
Superheater	Superheated steam	Exhaust gas
Evaporator	Water-steam mixture	Exhaust gas
Economizer	Water	Exhaust gas

Table 5
Averaged exhaust parameters for WHR design.

	Temperature (°C)	Flowrate (kg/s)
Deutz engines	443.1	19.52
Jenbacher engines	479.0	13.73
Mixture	460.3	33.25

and Jens and Lottes [34] the inner wall temperature T_{wall} and the heat flux q_{wall} can be estimated using the following correlations:

$$T_{wall} = T_s + (0.025 \cdot q_{wall})^{0.5} \cdot e^{-p_s(10/86.87)} \tag{15}$$

if $p_s < 50$ bar and $q_{wall} < 300,000\,\mathrm{W/m^2}$

The iteration is stopped when the evaporator's outlet temperature of the exhaust gas $T_{ex,eva,out}$ reaches the exhaust gas outlet temperature defined by the pinch point $T_{ex,eva,out} = T_s + pp$. This iteration process results in the number of pipes needed within the evaporator and therefore in the heat exchanger area of the evaporator.

2.3. Boundary conditions and restrictions

2.3.1. Ambient and calculating conditions

In order to achieve comparable results of all calculations, the same boundary and ambient conditions are used. For ambient conditions the thermochemical reference state [27] will be used:

- air temperature: 25 °C
- air pressure: 1.0 bar

Since it is not possible to install a water cooling at the power plant location (no sea or river water available), the organic fluid and steam respectively is condensed by an air cooled condenser after the steam turbine.

Furthermore, pressure and heat losses within pipes and heat exchangers of the bottoming cycles will be neglected.

2.3.2. Engine waste heat potential

The WHR design and dimensioning is based on the waste heat potential given by the ICEs. Averaged exhaust gas parameters, which were derived from the analyses of different sets of measurement data, are used to estimate this potential. The averaged parameters were calculated for the group of Deutz engines, the group of Jenbacher engines and the mixture of both engine type exhaust gas streams. The important exhaust gas parameters are outlined in Table 5.

In order to estimate the specific heat capacity of the exhaust gases, the exhaust gas composition is calculated with the measurements of the oxygen concentration within the exhaust gases and the assumption of complete combustion of the fuel. The results are given in Table 6.

The engine cooling water can usually be used for the condensate preheating both in the water/steam cycle and the ORC. In the case of the Ano Liosia power station, the cooling water of the GE Jenbacher engines is already being used in the biological treatment unit, so only the cooling water of the Deutz engines remains unused. However, according to [28] it is counterproductive to use a cooling water condensate preheater, since the condensate can also be preheated by the residual thermal energy of the exhaust gases after the HRSG. Thus, just the engine exhaust gases will be considered as heat input to the water/steam cycle.

Table 6
Averaged exhaust gas composition.

Composite (vol.%)	Deutz engines	Jenbacher engines	Mixture of both engines
CO_2	7.73	6.86	7.13
H_2O	12.48	11.08	11.77
O_2	6.75	8.35	7.60
N_2	73.04	73.71	73.49

Table 7
ORC working fluid properties.

	Pentane [36]	MDM [42]
Critical temperature T_c	196.6 °C	290.9 °C
Critical pressure p_c	33.7 bar	14.2 bar
Acentric factor ω	0.25	0.53
Temperature of decomposition	–	313 °C
Ignition temperature	260 °C	–
GWP	3	0
ODP	0	0

2.4. Bottoming cycles

2.4.1. Organic Rankine Cycle – ORC

One of the most common and widely applied methods for recovering exhaust gas waste heat is using the Organic Rankine Cycle (ORC) technology to generate additional electricity. Fig. 2 shows the working principle of the ORC. The working fluid is compressed with a pump, preheated in a regenerator and preheated and evaporated by the exhaust gas heat. The vapor is expanded in a turbine which drives the generator. The expanded vapor is then driven through the regenerator and finally condensed in an air cooled condenser (due to the lack of water adjacent to the power plant). The working media could be any organic fluid with the appropriate thermal properties in order to exploit efficiently the heat provided from the engines.

Two different types of ORC will be examined: a recovery system with (Fig. 2a) and one without the use of an intermediate thermal oil circuit (Fig. 2b), which transfers the heat from the exhaust gases to the ORC. Two major categories of the organic fluids will be assessed: the hydrocarbons and the siloxanes. These two fluid categories, whose physical and thermal properties are very attractive for such applications, are the most commonly used by ORC manufacturers. As representative fluids, pentane and octamethyltrisiloxane (MDM) were chosen. They have quite different thermodynamic properties; the most important ones for ORC design are mentioned in Table 7.

The thermal efficiency of ORC processes is increasing with higher critical temperatures of the working fluid, since exergy losses between the exhaust gas and the working fluid can be minimized due to lower temperature differences. However, in the case of MDM, which has the higher critical temperature T_c the acentric factor ω also becomes higher [35]. This means, that vapor pressure curve is more overhanging compared to pentane – the fluid is "drier". A higher grade of superheating after the turbine follows this fluid property. Fig. 3 shows the vapor pressure curve of both fluids in a T-s chart compared to water.

Both fluids have to be prevented from auto ignition or fluid decomposition. This safety issues can be solved by a thermal oil intermediate circuit. Despite causing additional losses in ORC thermal efficiency, this circuit is favorable especially in the case when exhaust heat from several engines has to be collected and transferred to one ORC unit.

Different cycle layouts, each with and without thermal oil intermediate circuit are concerned in this study. Two separate ORC systems, one for each section of the power station and the installation of one single ORC unit, using the intermediate thermal oil circuit to collect the waste heat from both groups of engines are examined. The cycle calculation is done by the thermodynamic simulation software IPSE Pro [24]. Important parameters for the simulations are given in Table 8.

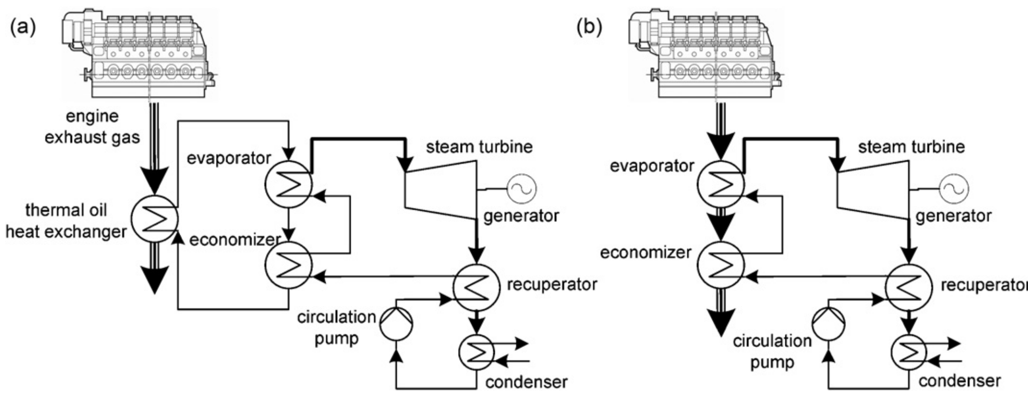

Fig. 2. Scheme of Organic Rankine Cycle (ORC): (a) with thermal oil intermediate circuit and (b) direct heat exchange.

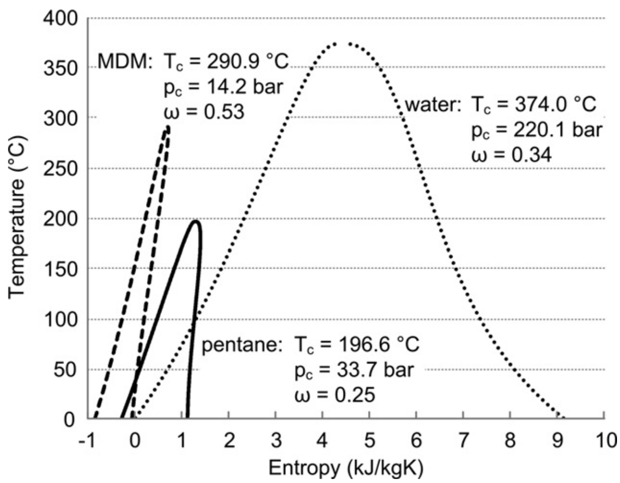

Fig. 3. T-s diagram of pentane and MDM compared to water.

2.4.2. Water/steam cycle

In the water/steam cycle the exhaust gases are recovered in a natural circulation heat recovery steam generator (HRSG). Saturated steam is produced by the preheater for deaeration with the low grade heat of the exhaust gases. The steam is not only used for heating the feedwater tank but as saturated low pressure (LP) steam for the LP-stage of the steam turbine as well. In the case of gas engines, the exhaust gases can be cooled down to about 90 °C, therefore a condensate preheater is installed within the exhaust gas flow. Fig. 4 shows the configuration of the water steam cycle.

Important parameters like live steam and condensate pressures are chosen according to the exhaust gas parameters and the ambient conditions (see Table 9).

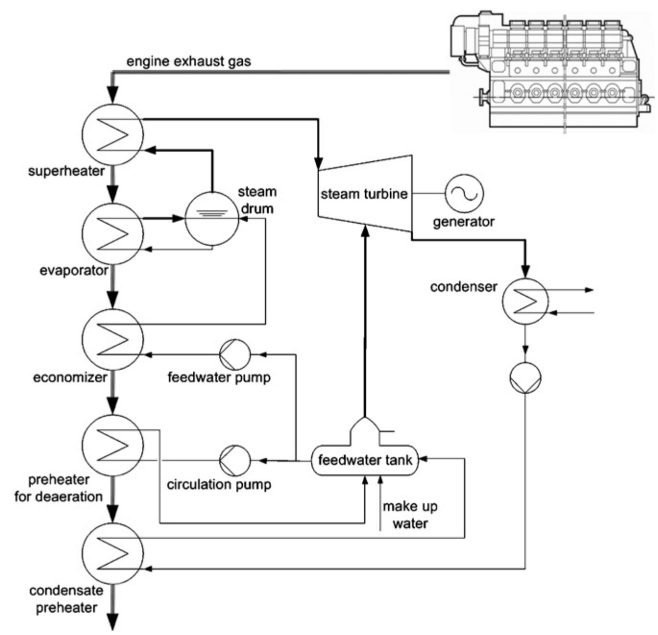

Fig. 4. Scheme of water steam cycle.

The thermodynamic calculations are performed using the REFPROP formulas of NIST [36] for water and steam properties within an Excel file.

As described in Section 2.4.1, one or two WHR units can be installed. In the first case one unit is bottoming all engines by collecting the engine exhaust gases in one HRSG. In the second case two HRSGs are installed – one for the Jenbacher engines and one for the Deutz engines – and the produced steam can be used in

Table 8
Design parameters of ORC configurations.

	Deutz	Jenbacher	Deutz	Jenbacher	Deutz	Jenbacher	All eng.
Working fluid	Pentane	Pentane	MDM	MDM	Pentane	Pentane	Pentane
Thermal oil circuit	x	x	x	x	-	-	x
Thermal oil pressure	3 bar	3 bar	3 bar	3 bar	–	–	3 bar
Thermal oil flow (kg/s)	17.9	16.2	18.3	10.8	–	–	31.9
Max/min thermal oil Temperature (°C)	275/100	250/95	300/115	350/110	–	–	275/100
Fluid pressure Before turbine (bar)	31	30	7	8	30	25	31
Fluid temperature Before turbine (°C)	192	189	245	254	190	180	192
Condensation pres. (bar)	1.5	1.5	0.2	0.2	1.5	1.5	1.5
Fluid temperature After recuperator (°C)	85	80	105	105	100	100	85

Table 9
Water/steam cycle parameters.

Parameter	Single cycle for Jenbacher engines	Single cycle for Deutz engines	Common cycle for both groups of engines
Live steam pressure p_s	30 bar	25 bar	25 bar
Live steam temperature T_s	420 °C	380 °C	400 °C
LP steam pressure p_{LP}	2 bar	2 bar	2 bar
Condensation pressure p_{cond}	0.15 bar	0.15 bar	0.15 bar
Pinch point temperature difference pp	15 K	15 K	15 K

either one or two steam turbines. It is also possible to install just one system for one of the groups of engines.

2.5. Results of thermodynamic cycle calculations

The results of the cycle simulations and thermodynamic calculations are presented in Figs. 5 and 6. These were calculated with respect to an averaged power output of the station of 18,864 kW (initial station el. power in Fig. 5) and an averaged electrical efficiency of 32.2% (initial station el. efficiency in Fig. 6).

Fig. 5 presents the increase of the total electrical power output of the station. Due to the higher waste heat potential, the WHR cycle for the Deutz engines produces the higher amount of electrical power for both the water/steam and the ORC system. Beyond this, two separated cycles for both groups of engines are producing more electrical energy than one common cycle for all the engines.

A Steam-Rankine system is more favorable in terms of power production and electrical efficiency compared to all variants of ORC systems. While the electrical power output can be increased by 2796 kW and 2618 kW by two separated and one common water/steam cycle respectively, the maximum power increase for an ORC system of 2264 kW can be calculated for two separated pentane cycles, which are directly heated without an intermediate thermal oil circuit. The ORC system with MDM as working fluid gives the worst results concerning electrical power (+1297 kW) and

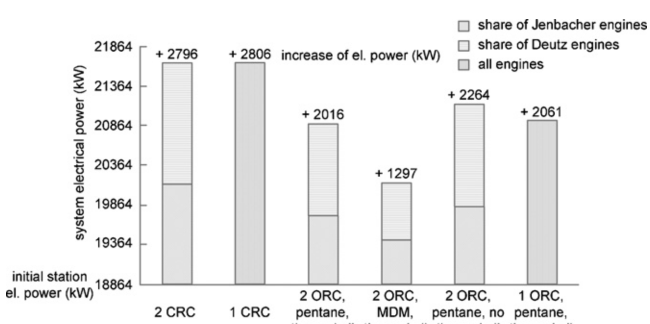

Fig. 5. Increase of the total power output of the station by different cycle configurations.

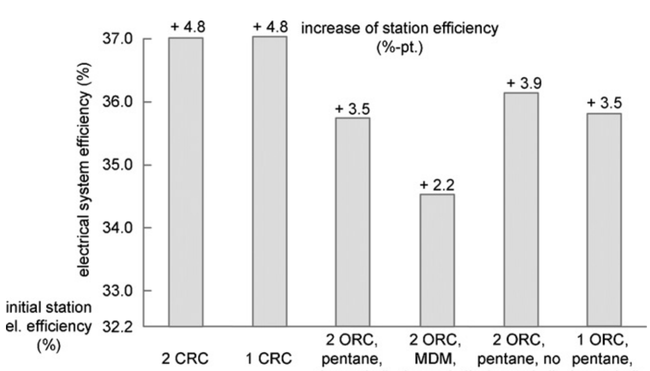

Fig. 6. Increase of electrical efficiency of the station by different cycle configurations.

efficiency (+2.2%-pt.). For this reason configurations of direct evaporation (without intermediate thermal oil circuit) of an ORC system with MDM as working fluid were not further simulated and are not included in the thermodynamic results.

The results of the electrical efficiency calculations are shown in Fig. 6. The total electrical efficiency of the station can be increased by 4.8%-pt. to about 37% by the two water/steam variants (+4.8%-pt. for both, two separated cycles and one common cycle), while an increase by more than 3%-pt. to about 36% is possible with all ORC variants which are working with pentane: +3.5%-pt. for two separated cycles with an intermediate thermal oil circuit, +3.9%-pt. for this configuration without the thermal oil cycle and +3.5%-pt. for one common cycle with thermal oil.

The reason for the better results of the water/steam cycle compared to the ORC cycles in terms of electricity generation and thermal efficiency is that the thermal energy of the exhaust gases can be better used with a water/steam cycle. This causes lower exergy destruction and exergy losses in the heat transfer process from the exhaust gases to the water/steam cycle than in the heat transfer process of the ORC configuration. In other words, the heat transfer efficiency η_{HT} of the available exhaust gas heat to the WHR cycle is higher (82.2% compared to 74.8% of ORC). In the ORC configuration the exhaust gases can be cooled down to about 120 °C while in the water/steam cycle they are cooled down to about 90 °C. Fig. 7 shows this fact in a T–Q diagram for the water/steam cycle (Fig. 7a) and the pentane-ORC (Fig. 7b) of the Deutz engines, where the area between the cooling curve of the exhaust gases and the heating curve of the working fluids is a measure for the exergy destruction and losses during the heat transfer process.

The economic analysis, which is conducted in Section 3, will reveal the most favorable configuration when both electricity generation and investment as well as O&M costs are considered.

2.6. Sensitivity analysis of water/steam cycle

In contrast to ORC systems, which are usually designed and constructed as complete standard modules, the water/steam cycle is constructed of single components, which can all be adapted and optimized for certain input parameters. Therefore it is advantageous to conduct a sensitivity analysis of the water/steam cycle performance for some relevant influence parameters. The effect of these parameters on the thermodynamic performance and on the necessary heat exchanger area within the HRSG, which is important for the following economic analysis, can be estimated. The HRSG is one of the most important components of the water/steam cycle and also the component which is the most vulnerable to changes in the design parameters of the water/steam cycle. Thus, the influence of three relevant design parameters – live steam temperature, live steam pressure and evaporator pinch point – on the power output and the heat exchanger area in the HRSG will be evaluated.

Fig. 8 shows the influence of the determination of the three selected design parameters on the power output, the necessary surface of the HRSG and the pressure loss of exhaust gas in the HRSG. The latter is important to avoid exhaust gas pressure losses greater than 20 mbar in order to avoid the negative influence of too high back pressures on engine performance.

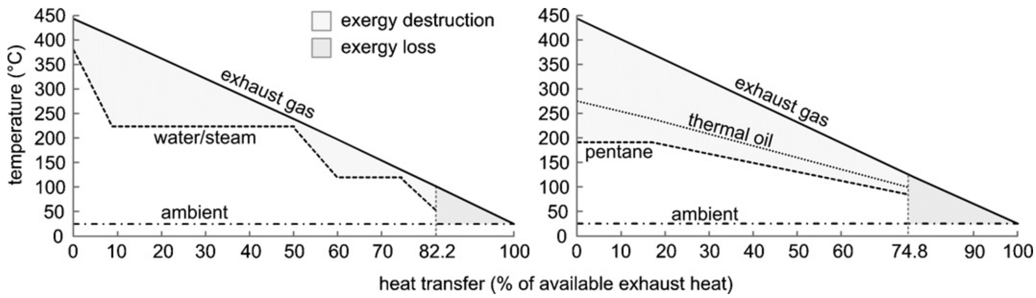

Fig. 7. T-Q diagram of (a) the water/steam and (b) the pentane cycle for the Deutz engines.

It is shown, that the heat exchanger surface area is strongly dependent on the pinch point determination (Fig. 8a). If the pinch point is reduced to 5 K (−66.6% compared to the original pinch point of 15 K), the HRSG area has to be increased by more than 20% while the cycle power output can just be enhanced by 2.1%. The exhaust gas back-pressure loss in the HRSG is proportional to the surface area and also increases by more than 20% with the pinch point reduction, but it is still far below 20 mbar (4.5 mbar). An increase in the pinch point to 25 K (+66.6%), decreases the area and the pressure loss by almost 9% and the power output by 2.1%.

If the live steam pressure is decreased to 15 bar (−50%), the electrical power decreases by more than 5.4% and the surface demand by 4.4%. An increase of the live steam pressure to a maximum of 80 bar (+166.6%), which is determined by the limitation of water-fraction at the steam turbine outlet (max. 10%), can increase the power output to 1302 kW (+3.1%). At the same time, the surface area is increased by 2.9%. The exhaust gas pressure loss is proportional to the surface area and far below the maximum allowable pressure loss all over the live steam pressure variation with a minimum of 3.5 mbar and a maximum of 3.9 mbar (−6.5% and +5.7% respectively on a basis of 3.7 mbar). Fig. 8b shows the relevant graphs.

The influence of the live steam temperature on the electrical power output is quite small. A variation of the live steam temperature from 380 °C to 460 °C (−9.5% to +9.5% of the initial temperature of 420 °C), varies the power output by −1.3% and +1.6% respectively. The HRSG surface area is decreased by −1.4% and increased by +5.8% respectively (see Fig. 8c).

3. Economic evaluation

The different cycle configurations are evaluated economically by means of their specific costs of electricity generation which are calculated for each WHR cycle.

3.1. Specific costs of electricity generation

The calculation of the specific costs of electricity generation is based on the static annuity method provided by VDI 2067 [37]. Therefore, costs are subdivided into three categories:

- capital-related
- consumption and requirement related
- operation related

To simplify calculations only the main costs are considered, which include the following:

- investment costs as capital-related costs,
- costs for operation (salaries of operators), maintenance and repair as operation related costs.

Since water/steam cycles as well as ORCs run on the exhaust gas heat of an engine, there are no consumption related costs since no additional fuel is needed (costs for lubrication oil, chemicals for water treatment, losses of working fluid, etc. are neglected). The static method of calculation is used, thus a price change factor will not be considered. The interest factor, which is assumed for investments in energy production facilities, is 10%. The annuities of capital-related and operation-related costs are calculated with the equations given by VDI 2067, as follows:

Annuity factor a: required to allocate singular payments (e.g. the investment costs A_0) to yearly amounts that are all equal throughout the period of observation *T*.

$$a = \frac{(q+1)^T \cdot q}{(q+1)^T - 1} \tag{16}$$

Annuity of capital-related costs A_c: the equalized yearly payment, that is calculated with the annuity factor *a*, the investment costs

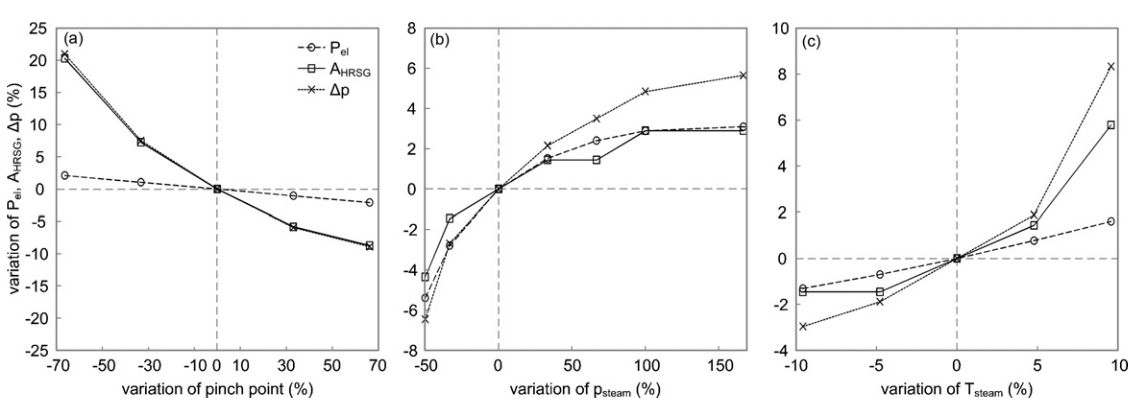

Fig. 8. Influence of (a) pinch point variation, (b) live steam pressure variation and (c) live steam temperature variation on HRSG area, power output and pressure loss in HRSG at the example of an HRSG for the GE Jenbacher engines.

A_0, the costs for replacement purchases of components $A_1 \ldots A_n$ and the residual value R of these components. The annuity is calculated separately for the main cycle components, with the following equations:

$$A_c = (A_0 + A_1 + A_2 + \cdots + A_n - R) \cdot a \tag{17}$$

$$A_i = \frac{A_0}{(q+1)^{i \cdot T_l}}; \quad i = 1 \ldots n \tag{18}$$

$$R = A_0 \cdot \frac{T_l - T}{T_l \cdot (1+q)^T} \tag{19}$$

Annuity of operation-related costs A_{op}: the yearly payment that is necessary for operation and maintenance of a water/steam or ORC cycle:

$$A_{op} = f_m \cdot A_0 + S \tag{20}$$

The expenses for operation and maintenance f_m in percentage of total investment per year are given by VDI 2067 for each component. Typical life times of components T_l are also given by VDI 2067. Due to the few moving parts of these systems, the availability factor is high and equal to about 97%. Taking into consideration that all the calculations are based on the real every day operation of the station and not the nominal, the full load hours $T_{fullload}$ of the WHR systems are about 8500 h annually. It is assumed that one additional skilled worker is needed at the power station for the operation of the WHR system. The salary S of this worker is 35,000 € per year.

The specific costs of electricity production are then calculated by:

$$c_{EP} = \frac{A_c + A_{op}}{P_{net} \cdot T_{fullload}} \tag{21}$$

Some economical factors, which are important for the calculation of specific costs of electricity generation will be used as following for all cycle configurations:

- interest factor: 10%
- factor for maintenance: 0.5% (ORC) and 1.0% (water/steam) of initial investment costs per year; the higher factor for the water/steam cycle accounts for the more complex system of a water steam cycle, which is build of individual components and not as complete modules like ORC units
- service life of installation and lifetime of components: 20 years
- observation period T: 20 years.

3.2. Water/steam cycle

As explained in Section 2.6, the components of the water/steam cycle have to be ordered and installed individually since complete installations like in the case of the ORC systems are usually not offered. Likewise, the investment or capital-related costs will be calculated individually for the main components.

Fig. 9 shows the dependence of the installation costs of the three main components on the size of the system (expressed in terms of steam turbine power). The HRSG and the condenser can be built from a number of standardized modules according to the demanded power, such the specific costs of these modules are just weakly correlated to the total size and power of the system. The HRSG specific investment costs range from 420 €/kW_{ST} to 380 €/kW_{ST} and these of the air cooled condenser from 220 €/kW_{ST} to 130 €/kW_{ST}. However, the steam turbine is strongly dependent on the total power output ranging from 410 €/kW_{ST} for a 6.5 MW steam turbine to 200 €/kW_{ST} for a 24.5 MW steam turbine. The data used for these calculations was taken from more than 20 OEM offers for water/steam cycle systems and components.

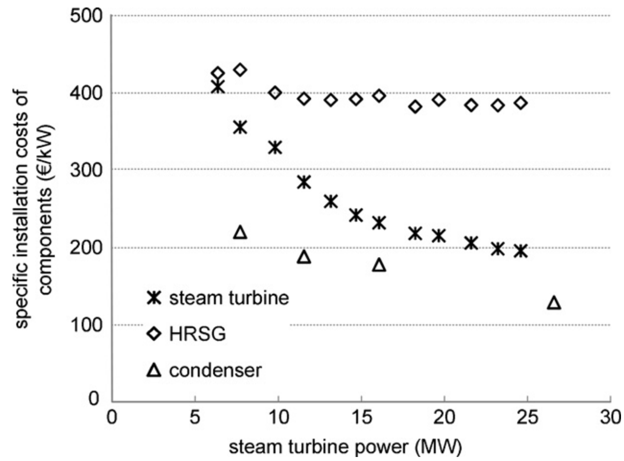

Fig. 9. Installation costs of main components of the water/steam system.

The price for the HRSG has to be calculated as well to a specific price per HRSG heat exchanger area in order to evaluate the influence of the main design parameters on the investment costs of the water/steam system. Therefore again some offers from HRSG manufacturers were used. These reveal a price of about 70 €/m^2 for the heat exchanger area, which is about 60% of the total HRSG costs. If the costs for piping, drums and valves are included and correlated to the heat exchanger area, a specific price of roughly 120 €/m^2 for the total HRSG can be calculated. In order to account for increasing prices of materials with increasing pressure levels, the HRSG area price will be increased by 12% per pressure class (PN 25, PN 40, PN 63 [38]) starting from the basic price of 120 €/m^2 for the pressure class PN 25.

The investment costs for the turbine and the air cooled condenser will be extrapolated from the data available to the requested steam turbine power. In order to calculate the total investment costs of the system, 20% of the costs of the three main components will be added to the total costs in order to account for costs of e.g. piping, pumps, auxiliary equipment, installation and commissioning. Table 10 compiles the installation costs of the three basic water/steam cycle installations.

3.3. ORC

In the case of the ORC configuration, there are several companies worldwide that offer integrated solutions for these applications. For the implementation of this solution on the present station, several different companies were asked to offer a complete and real economic proposal for a preliminary evaluation of the project. For this reason, three different offers will be examined, from Company 1, Company 2 and Company 3.

Company 1 proposed to use one single ORC unit with an intermediate thermal oil circuit which will collect the waste heat from both groups of engines. According to the company, the estimated energy production of the system would be 1801 kW el and the

Table 10
Installation costs of the water/steam cycle.

	Jenbacher engines	Deutz engines	Jenbacher + Deutz engines
Electrical power P_{el}	1263 kW	1533 kW	2806 kW
Installation costs:			
- HRSG	692,160 €	764,160 €	1,324,080 €
- Steam turbine	685,414 €	818,546 €	13,88,189 €
- Condenser	323,176 €	389,480 €	690,041 €
- Auxiliaries	340,147 €	394,437 €	680,462 €
Sum	2,040,885 €	2,366,623 €	4,082,772 €

cost 1,655,000 €. Moreover, the transportation and installation cost would be 60,000 € and the cost for the thermal insulation of the unit 31,000 €. Apart from these, the cooling system along with the pipes and the pumps of the circuit costs about 650,000 €, the connection to the grid costs 100,000 € and the civil works for the unit and the control room about 100,000 €. All these considered, along with the cost of the intermediate thermal oil circuit, the total cost of this offer is about 3,200,000 €. Concerning the operation related costs, due to the few moving parts, the maintenance and repair works of the unit require only 3–5 h weekly and they are offered by the company at the cost of 15,000 € annually.

Company 2 is mainly focused in smaller scale applications and designs only small ORC units. As a result, in order to exploit the high waste heat output, the company proposed the installation of 9 small units of 125 kW each (nominal load), with a total output of only 917 MW. The total cost of the design, manufacture, installation and connection to the grid of the 9 units which will be placed into separate containers is 4,300,000 €. In this case, the maintenance of each unit costs 8000 € annually.

Company 3 proposed to use one single ORC unit for both groups of engines. An intermediate thermal oil circuit will collect the heat from the exhaust gases of the engines through 15 small heat-exchangers, one in each chimney. According to the company, the energy production will be about 2000 kW while the unit costs 3,200,000$. Considering the current euro-dollar ratio, along with the cost of the intermediate thermal oil circuit, the connection to the grid and the transportation and installation of the unit, the total cost is about 3,000,000 €. The operation related costs are again very low due to the high reliability of such units and are estimated to be about 15,000 € annually.

3.4. Results of economic calculations

Fig. 10 shows the costs of electricity generation, which were calculated for the different systems in €/MW h. It can be seen, that they are very similar for all water/steam and ORC systems, ranging from 23 €/MW h to 28 €/MW h, with the exception of the ORC offered by Company 2. The reason for the much higher costs of this ORC is that the company can just offer small modules with a power of 125 kW. Therefore nine units are needed in order to recover all the available exhaust gas heat.

The most competitive solutions are the installation of one water/steam unit for all engines with electricity production costs of 23.29 €/MW h or an ORC offered by Company 3 with 23.67 €/MW h.

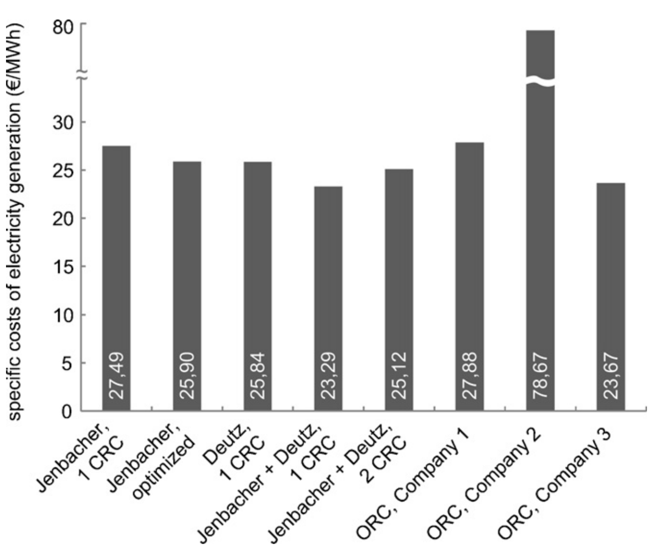

Fig. 10. Costs of electricity generation for various configurations.

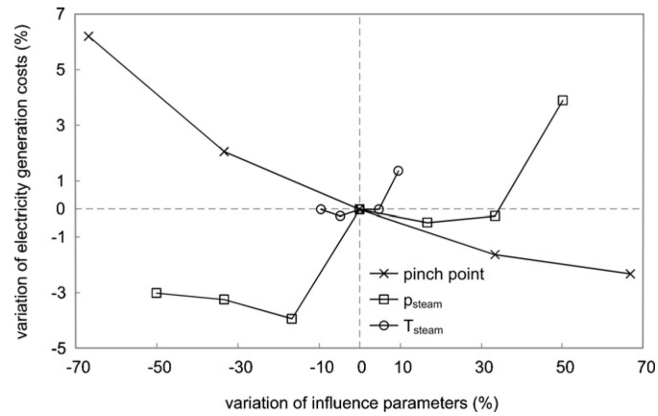

Fig. 11. Variation of costs of electricity generation with the variation of important influence parameters (pinch point, live steam pressure, live steam temperature).

A clear trend towards water/steam cycle or ORC cannot be seen in this range of WHR power output or exhaust gas temperature levels. Therefore, the decision for the most favorable system can only be based on precise offers for this specific application, taking into account not only the costs of installation but also detailed costs for operation, repair and maintenance, which could only be roughly estimated by factors in this study.

3.5. Sensitivity analysis of water steam cycle costs

The influence of important cycle parameters on the economic performance of the water/steam cycle for the GE Jenbacher engines is shown in Fig. 11. It can be seen, that the pinch point and the live steam pressure level have a great influence on the costs of electricity generation. They can be decreased by about 2.1% if a pinch point of 25 K instead of 15 K is selected for the evaporator. The graph of the influence of the live steam pressure shows very well the increasing costs for the piping material and boiler equipment with increasing pressure levels (staging of the graph). A decrease of the electricity generation costs by 3.8% can be achieved, if the live steam pressure level is lowered to 25 bar. The live steam temperature has just a small influence with a decrease of about 0.2% in electricity generation costs for a live steam temperature of 400 °C instead of 420 °C. Fig. 10 shows that the costs of electricity generation of an optimized water/steam cycle can be decreased by about 6% to 25.9 €/MW h.

However, the influences of the different parameters were evaluated while being isolated from each other, with the variation of one parameter while keeping constant the other two. But since the HRSG performance is depending on all parameters simultaneously, an approach for multi-parameter optimization (e.g. genetic algorithms) might reveal even better results [39].

4. Conclusion and outlook

In this study the possibilities for the efficient use of the waste heat of landfill gas-fired internal combustion engines were examined. Two waste heat recovery cycle alternatives were proposed – the water/steam cycle and the Organic Rankine Cycle. Different configurations of these cycles were modelled within software environments suitable for thermodynamic calculations and afterwards evaluated both under thermodynamic and economic considerations.

The thermodynamic analysis of all WHR cycle alternatives shows that water/steam cycles generally reach a higher increase of the electrical station efficiency of up to 37%, compared to ORC alternatives. With the latter, a station efficiency of about 36% can be reached with a pentane cycle without a thermal oil intermediate

circuit. However, applying this kind of cycle could cause high risk levels, since pentane is very flammable (compared to e.g. MDM). Therefore most companies working with pentane cycles use thermal oil circuits.

The use of all the exhaust gases of the machines is technically difficult. In the case of an ORC configuration, a secondary oil circuit is used, which is more practical whereas the use of one heat recovery steam generator for a water steam cycle may be more efficient, but requires complicated engineering in the existing plant.

Since the decision of an investment in a WHR cycle can not only be based on thermodynamic considerations, an economic analysis was also conducted. This reveals that, despite of the thermodynamic advantages of the water/steam cycles, both cycle alternatives reach the same level of costs of electricity generation of about 23 €/MW h, which makes the consideration of detailed offers for this specific application extremely important.

However, the sensitivity analysis of the water/steam cycle configuration for the Jenbacher engines shows, that the costs of electricity generation can be decreased by up to 6%, if design parameters are optimized. This parameter optimization can only be conducted for the water/steam cycle, since all components are designed individually compared to ORC systems, which are offered as turn-key modules by the OEMs. Thus, a detailed parameter optimization of all proposed water/steam cycle alternatives might bring clear economic advantages for the water/steam cycle, compared to the ORCs. This could be the scope of further studies.

Another object of further studies should also be the evaluation of the engine combined cycle systems under transient operational conditions [40,41]. The Ano Liosia power plant is working at an average load of 80%, which means that all engines are working under transient and different loads. The impact of this on the operation and power output of the WHR system should be studied.

Acknowledgements

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